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Energy Saving Potential in Existing Volumetric Rotary Compressors

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Abstract

The issues of energy and Carbon saving in energy intensive sectors, along with that of energy generation from renewable sources, have been recently receiving a growing awareness, as they are perceived as the most effective ways to deal with global sustainability commitments. The Compressed Air Sector (CAS) accounts for a 10% worldwide electricity consumption, and thus is being re-thought as an area offering great opportunities for improvement. Considering that the compression is responsible for a 10-15% consumption, it is vital to pay attention to machines performances. An overview of present compressor technology is given and saving directions for Screw and Sliding Vanes machines are analysed: interesting source of information was the Compressed Air and Gas Institute (CAGI), whose data have been processed, in order to obtain consistency with fixed reference pressures, and organized as a function of main operating parameters. The overall efficiency has been split and all different sub-terms (adiabatic, volumetric, mechanical, electrical, organic) considered separately. This has allowed a term-by-term evaluation of the margin for improvement. The heat recovery from the oil into mechanical energy via an Organic Rankine Cycle (ORC), together with the thermodynamic improvement during compression phase opens the way to a step change concerning the specific energy consumption.

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Keywords: compressed air systems, sliding vane rotary compressor, positive displacement compressor, dual stage, waste heat recovery, Organic Rankine Cycle

1. Introduction

During the last decade, the energy issue has been receiving attention, more than ever before, mainly because of the growing global instability that has made future energy scenarios lees predictable than in the past. Amongst the main drawbacks, those that the market seems to suffer the most, are the impossibility to define a shared energy policy and the difficulty in establishing a globally accepted agenda on energy planning. The current energetic economy is dominated by fossil fuels (oil, gas and coal, 85%), with hydro, nuclear and other renewables accounting for the remaining 15% [1]. As a result, the CO_2 amount into the atmosphere is growing at a rate that, considered the present concentration of 396.16 ppm [2], will soon frustrate the attempt to meet the political target of 450 ppm by 2035, fixed by the Intergovernmental Panel on Climate Change as a safeguard limit. This unbalance between fossil and

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sustainable development suggests that, at least in the near future, energy efficiency and energy recovery will be, most likely, the only measures left to take.

Since the paper focuses on CAS, whose energy demand entirely concentrates on electric energy, the overall electricity consumption for different Countries is worth consideration. Data from the International Energy Outlook 2011 give evidence about the fact that a great part of the cumulative electricity demand comes from industry in both developed and developing Countries [3]: percentages up to 70% can be observed, with BRICS showing the highest values. What significantly changes, among developed and developing Countries is the proportion in sector consumption: absolute values show that in EU, 40% electricity demand comes from industry, whereas in US, this share ranks at 25%; greater shares characterize the electrity demand in BRICS Countries (e.g. 70% chinese electricity consumption is industry related). Such a scenario gets even more unbalanced when the projections for future electricity consumption are taken into account and 2020 is assumed as reference time horizon: in the 2010-2020 decade, a linear increase in yearly electricity consumption is expected from the present 16000 TWh/yr up to 24000 TWh/yr, leading to a cumulative 200000 TWh consumed. Considering that compressed air production accounts for a mean value around 10% worldwide overall electricity consumption [4] and that the mean specific CO_2 emission rate per kWh is 600 gCO_2/kWh, a synthesis of the CAS energy and Carbon dimension is possible: consolidated data in 2010 demonstrate 0.9 $GtCO_2/yr$ and predictions in 2020 bring to 1.44 $GtCO_2/yr$. In terms of Carbon, during the decade 2010-2020, the cumulative emissions could be estimated equal to 2 $ppmCO_2$. This is a 4% overall admitted CO_2 increase in atmosphere, if the stabilization at 450 $ppmCO_2$ is considered a political goal.

Nomenclature

η	efficiency
Δ	variation
h	specific enthalpy $[J/kg]$
$\tilde{R}$	individual gas constant $[J/(kg\,K)]$
T	temperature [K]
c_p	specific heat at constant pressure $[J/(kg\,K)]$
k	adiabatic isentropic exponent
γ	polytropic exponent
β	compression ratio
q_s	specific power consumption $[kW/(m^3/min)]$
m	volumetric flow rate $[m^3/min]$
R	Pearson's product moment correlation coefficient
A, a	regression model parameters
$_{glob}$	global
$_{ad,is}$	adiabatic isentropic
$_{vol}$	volumetric
$_{mech}$	mechanical
$_{org}$	organic
$_{el}$	electrical
$_{real}$	real
$_{inl}$	inlet
$_{msr}$	measured
*	reported to reference pressure levels
**	interpolating

2. Energy Analysis of Industrial Compressors

The reason why the CAS is addressed as an interesting sector of application of technological improvement, when CO_2 reduction is considered as a major future concern, is twofold. Firstly, it accounts for a 10% worldwide overall electricity consumption. Secondly, energy saving/recovery appears very promising, since compressed air systems show a high level of complexity in spite of very low characteristic efficiencies. In detail, compressor technology and operation accounts for a mean 10-15% consumption, while pressure losses in pipes, leakages and wasteful end uses account for a greater share [4].

Many independent studies [4–6] focus on the main criticalities in present CAS and provide estimations of the energy saving potential for a great variety of interventions: end use devices optimization, air leaks reduction and use of sophisticated control systems appear to be the most effective actions (with potential saving of 30%, 20% and 15% package power, respectively), while the compressor upgrading follows, with lower saving potential (5-10% package power). The opportunity offered by the CAS thermal management (waste heat re-use, innovative cooling techniques definition during compression) still remains an interesting matter of development. In order to evaluate the actual potential of such interventions, many factors (e.g. the impact on consumption of compressor technology and control strategies, the energy and capital costs) must be considered, even though the opportunities offered by the compressor performance improvement are not in question. In [7,8], the benefits produced by investing money in more efficient compressors and those coming from a reduction in electricity costs are correlated one other. A 10% investment increase is justified only if the saving is greater than 4% with respect to present technology. Since the ratio between the two parameters remains almost constant, serious consideration should be paid to invest in more energy efficient compressors when the additional capital cost can be easily offset by the cost of the energy saved. This suggests the need for machines more oriented to reduce compression-related consumption.

As a consequence, it is vital to get deeper inside the transformation issue (from electrical energy to compressed air) and to perform an energy analysis of those compressors, that represent the technological standard in industrial applications. Particularly, rotary volumetric machines, and among them, Sliding Vanes (SVRC) and Screw types are analyzed, as the most common and widespread technologies.

The evaluation of the potential energy saving during compression requires the definition of ideal reference transformations. Two transformations appear particularly fitting the purposes of the analysis: an adiabatic isentropic, which is the best approximation for real compression and an isothermal, reference transformation with the minimum work required. According to a common approach, the overall system efficiency can be seen as:

$$\eta_{glob} = \eta_{ad,is}\,\eta_{vol}\,\eta_{mech}\,\eta_{el}\,\eta_{org} \tag{1}$$

In order to evaluate the potential savings, each term in Equation 1, deserves some attention:

- $\eta_{ad,is}$: it accounts for the thermodynamics of compression and it is expressed by considering an adiabatic (isentropic) transformation as reference. The reason for such a choice is that it closely reproduces the one stage compression that takes place in present machines, both Screw and Sliding Vane. Real work can be evaluated from the indicator diagram, which demonstrates that: a) the induction process, in both SVR and Screw compressors can be considered as steady, while for the exhaust such assumption is very far from being verified: even though pressure pulsations occur, they do not deviate significantly from a mean isobaric transformation equal to the line pressure. An optimized eccentricity and port geometry can reduce those oscillation [9]; b) following present technology, the compression when the vane is closed is very close to an adiabatic (isentropic) compression [10]. Therefore, according to experimental data, it is possible to preliminarily fix this efficiency close to one. From these considerations, a potential greater improvement can be done only improving the closed volume transformation moving it far from being adiabatic. In particular, an isothermal transformation would introduce a relevant reduction in specific power requirement, but it would require a different compression technology. Recent advancements in SVRC aim at decreasing compression work, by cooling the air during compression, injecting a fine oil spray in the vanes: in such case, care must be done in order to avoid the presence of the oil in vapor phase which would re-increase the compression work [11,12]. So, Δh_{real} in Equation 2 can be evaluated according to the indicator diagram, but also evaluating the enthalpy difference of the air between inlet and outlet, being actual machines, as already observed, characterized by an isobar filling ad emptying process and a closed volume transformation close to an adiabatic. In fact, in standard set-up for compression transformation in rotary volumetric machines, air leaves the machine at a temperature much lower than the adiabatic value. As known, this is due to the strong cooling by the abundant oil present inside the machine. So, the overall transformation can be seen as a polytropic (close to the adiabatic), followed by an almost constant pressure cooling process. The adiabatic isentropic efficiency can be expressed as:

$$\eta_{ad,is} = \frac{(-W)_{ad,is}}{(-W)_{real}} = \frac{\Delta h_{ad,is}}{\Delta h_{real}} = \frac{\tilde{R}\, T_{inl} \left[\beta^{\frac{k-1}{k}} - 1\right]}{c_p \; T_{inl} \left[\beta^{\frac{\gamma-1}{\gamma}} - 1\right]} \tag{2}$$

- η_{vol}: it takes into account that some air is not delivered after compression, mainly because of losses in the filling and emptying processes: while intake can be considered as a steady process [13], when air is exhausted an intrinsic unsteady behavior is more or less always present, mainly because the line pressure differs from the one inside the machine. The vane is squeezed during rotation, but a residual quantity of air inside it cannot be expelled, resulting in a recirculation among vanes during the compression phase. It is known that, according to compressor type, margins for improvement are still possible in SVRC - mainly reducing air leaks on the vertical planes, where sealing effect is not guaranteed - and in Screw machines - on the so called ”blow hole line” [14]. The effect of rotational speed on volumetric efficiency is worth being discussed. In SVRC, there are no operational constraints on its value: a stable oil film between surfaces in relative motion prevents internal pressure-controlled leaks through clearances, allowing to fix this efficiency close to one. In Screw compressors, low rotational speeds increase the air recirculated through the blow hole line, while they have residual effects on the leaks through the screws/compressor housing clearances and between meshing rotors [15]. In best in class machines, the comparison between ideal and rated flow rate demonstrates that this efficiency is high and close to one;
- η_{mech}: it gives the greatest contribution to the efficiency shift from the unit, accounting for friction between parts in relative motion. Room for improvement is still possible in both SVRC (i.e. reduction in friction between blade tip and slot, rotor and stator and even if negligible with respect to the previous contributions, friction in bushings) [16] and Screw machines (i.e. reduction in friction between rotor and stator, driving and driven rotors, motion transmission and speed reduction gears and friction in bearings) [17]. In SVRC, contact surfaces are at the blade tip-stator interface (the blade is expelled by centrifugal force) and between blade and slot surface (blade tilting because of pressure difference between adjacent vanes). The variation of the geometrical diameter-length ratio appears the most promising strategy to reduce friction, since it allows to modify reaction forces on the blade: higher stator diameters lead to higher centrifugal forces, whereas an increase in rotor diameter requires lower eccentricity, with lower relative speed between blade lateral surfaces and the slot, decreasing losses. Frictional losses reduction in SVRC can be achieved with rotational speed modulation too. In particular, losses reduce at lower speeds: in these conditions, the compressor length has to be increased to keep the mass flow rate constant, with a linear increase in the blades mass. Centrifugal force receives a positive contribution from this mass increase, although the change is driven by the downspeeding (linear dependence on the mass vs. quadratic dependence on the rotational speed). Friction in Screw compressors depends on bearings accuracy and rotors accurate shaping. Due to the principle of operation of Screw compressors, contact during rotation cannot be avoided and contributes to wear and surfaces damages at the rotor-rotor contact areas, thus increasing loss in mechanical power [18]. Rotational speed reduction certainly decreases friction but it is detrimental on volumetric efficiency: its potential in Screw machines is then well below the one in SVRC;
- η_{el}: prime mover is usually an electric motor. Its inefficiencies affect the package power requirements. Electric motor improvements have been proposed in CAS and are on the market;
- η_{org}: it takes into account the power absorption by auxiliaries, like the fan that serves the radiator, when oil is cooled by air, or the power absorbed by the pump, when the radiator is liquid cooled. In reality, the power needed to pressurize the oil inside the machines can be considered as power absorbed by an auxiliary too. Usually, the oil quantity is abundant for both SVRC and Screw machines and even though the high oil density, with respect to the air, should make the power requested for oil pressurization negligible, the abundant oil flow

rate increases this contribution. In present machines, it reaches 5-7%: so, the only way to improve organic efficiency would be the oil flow rate reduction. In SVRC this has been preliminarily done [19], but sealing and friction should be taken under control.

So, in Screw and SVRC the global efficiency mainly depends on: a) the mechanical, organic and electrical efficiency, whose improvement appears an effective direction, with the mechanical representing a much more interesting matter of development, b) the thermodynamics of the compression during closed volumes.

3. Industrial Compressed Air Systems Performance Analysis

An analysis of the energy consumption of existing machines has been done. Interesting source of data has been the CAGI, where data sheets of different manufacturers compressors are available. A clear overview of the energetic performances of present compressor technology came out from the processing of thousands of data sheets. Even if available in a standardized form and with an unified structure, data required a deep and time consuming treatment, in order to avoid inconsistencies in comparing performances. Main reason for this, is the difference between measured pressure levels and those considered as references. To allow a comparison between machines at the same delivering pressure, the following procedure has been applied:

1. reference pressures are fixed equal to 8, 9, 10 and 11 bar: they are the most common delivered pressures;
2. if the pressure delivered by a specific machine remains in the 5% of the reference value assumed, data are processed and modified in order to refer to reference pressure levels; otherwise, they are excluded;
3. for the processed data, reference and measured pressures, along with the compression ratios, are known. Energy consumption can then be referred to common pressure reference values, by observing that, the volumetric, mechanical, electric and organic terms in Equation 1 can be considered as constants, when the energy consumption from the measured values is referred to reference values. For this reason:

$$\Delta\eta_{glob} = \Delta\eta_{ad,is} \tag{3}$$

Where $\eta_{ad,is}$ is calculated according to Equation 2. Then:

$$\frac{d\eta_{ad,is}}{d\beta} = const(k, \tilde{R}, T_{inl}, c_p, \gamma)\,\beta^{-\frac{1}{k}} \tag{4}$$

The specific consumption variation related to the global efficiency and the pressure ratio is given by:

$$\frac{\Delta q_s}{{q_s}^{msr}} = \frac{{q_s}^{msr} - {q_s}^{*}}{{q_s}^{msr}} = -\frac{\Delta\eta_{glob}}{\eta_{glob}} = -\frac{\beta^{*} - \beta^{msr}}{(\beta^{msr})^{-k}} \tag{5}$$

The specific consumption that corresponds to the compression ratio assumed as reference is then:

$${q_s}^{*} = \left(1 - \frac{\Delta q_s}{{q_s}^{msr}}\right){q_s}^{msr} \tag{6}$$

while, for the mass flow rate:

$$m^{*} = m^{msr} \tag{7}$$

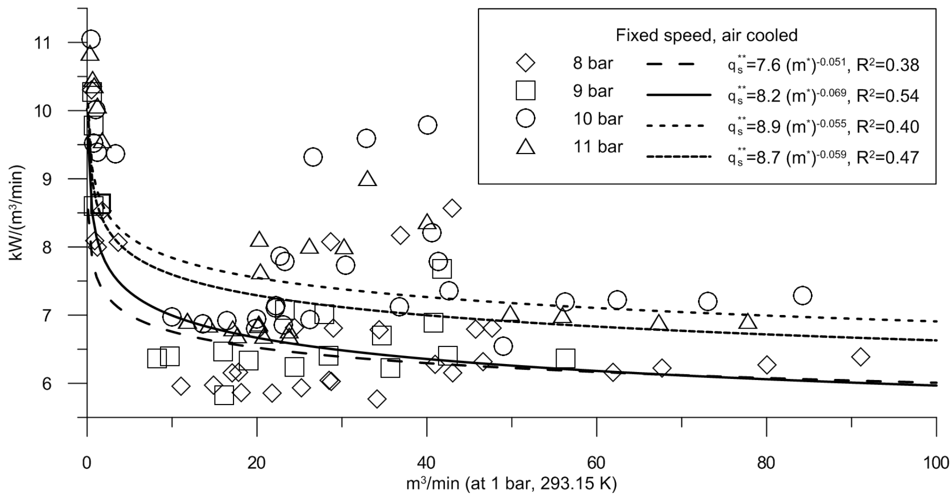
It has to be noted that such an approach applies in case of small difference between measured and reference pressures, so that the assumption on the nature of the transformation will not affect the calculations. In reality, rotary volumetric machines show an isochoric phase immediately after the port opening, as pressure inside the vane is different from line pressure, quite far from the assumption of a polytropic. Applying the correction procedure, almost 25% of the original data has been excluded, since the distance between measured and reference pressure was outside the 2.5% range. Figure 1 shows the performances of market compressors, when all the data are recalculated at reference pressures: for sake of privacy, they are presented without mentioning the manufacturers name. Data have been organized as a function of pressure delivered at rated (usual design operation) and zero (frequent in case of load/unload control) flow and as a function of the oil cooling technique (air or water). The following considerations apply:

- specific power decreases in general with respect to flow rate, so bigger machines are more efficient: for air cooling, at 5 m^3/min, 20 m^3/min and 50 m^3/min flow rates, the mean specific power is 7.7 $kW/(m^3/min)$, 7.0 $kW/(m^3/min)$ and 6.5 $kW/(m^3/min)$, respectively; for the same flow rates, water cooled machines show specific power consumption of about 7.0 $kW/(m^3/min)$, 6.7 $kW/(m^3/min)$ and 6.5 $kW/(m^3/min)$, respectively. This trend is particularly evident for air cooled machines and for higher pressures delivered;
- machines, in which the oil is cooled by water, show higher efficiencies than those in which oil is cooled by air, since a pump (circulating an incompressible fluid) always has a negligible impact on package power demand, with respect to a fan;
- for a delivery of 8 bar, air cooled machines have a significant technological scatter: for a rated flow rate of 10 m^3/min, the mean value is 6.8 $kW/(m^3/min)$, with a 1.2 $kW/(m^3/min)$ scatter, meaning that huge saving in energy absorption can be achieved by properly choosing the compressor. When flow rate increases, specific consumption shows an asymptotic trend, with lower mean specific consumption and scatter (for a flow rate of 50 m^3/min, the mean value is about 6.2 $kW/(m^3/min)$ and the scatter is 0.5 $kW/(m^3/min)$). The scatter for higher flow rates is close to zero. For water cooled machines, the scatter is an order of magnitude lower than the corresponding in air cooled, resulting in a negligible effect on performances;
- for higher delivered pressures, the differences among air and water cooled increase, inviting to water cooled version. For a delivered pressure of 9 bar, the scatter further reduces, leading to a compressors population that is tightly distributed around the performances mean value: for a flow rate of 30 m^3/min, the mean value is around 6.6 $kW/(m^3/min)$, with a scatter of about 0.4 $kW/(m^3/min)$. The previous considerations keep their validity when referred to water cooled compressors, as demonstrated by the negligible scatter that characterizes performances yet for a 20 m^3/min flow rate;
- the technology is not aligned to best standards: a reason for this is the massive market presence of machines, designed years ago, with no evidence of any upgrading effort on them. Equation 8 reports the expression of the coefficient of determination R^2, a statistical measure of how well the regression model (in this case: $q_s^{**} = A\, m^{*-a}$ approximates energetic performances of existing machines: low R^2 mean that technology does not meet the best standards expectations.

$$R^2 = \left(\frac{\sum(m^* - m^{**})(q_s{}^* - q_s{}^{**})}{\sqrt{\sum(m^* - m^{**})^2 \sum(q_s{}^* - q_s{}^{**})^2}} \right)^2 \tag{8}$$

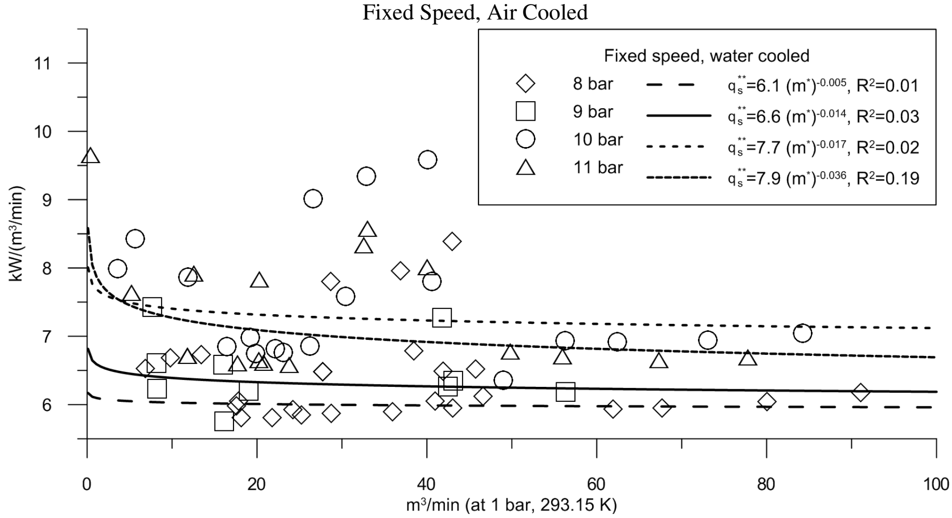
The potential energy saving during compression is given by the distance between the electric specific consumption of real machines and the one calculated for ideal compressions inside ideal machines (i.e. η_{glob}=1) (Figure 2). In order to set up a minimum energy saving, only premium machines (i.e. those with the minimum energy requirement for any compression ratio) are considered. As previously observed, the specific consumption shows an asymptotic trend with increasing flow rate, so premium machines are usually big size compressors; this also means that the energy saving is evaluated with reference to compressors with very little spread in performances.

The higher spread (i.e. the lower R^2) and the higher specific consumptions of the great majority of market compressors, assure that in mean situations a higher energy saving is probable. Dots in Figure 2 represent the market best in class machines and lines represent an adiabatic isentropic and an isothermal transformation, with a global efficiency equal to one. Real data points seem to be closely fitted by an adiabatic transformation, with an efficiency no longer equal to one, with the main contribution to its shift coming from mechanical, organic and electrical terms. Indeed, fixed speed machines are very well fitted by an overall global efficiency close to 0.79 if oil is cooled by air and 0.81 if oil is cooled by water: such a difference mainly depends on the organic efficiency, that is lower when oil is cooled by air, since the power absorption by the fan is greater than the one absorbed by the pump that serves the radiator in those machines in which oil is cooled by water. The distance between the adiabatic which interpolates the best



(a)

Fixed Speed, Air Cooled

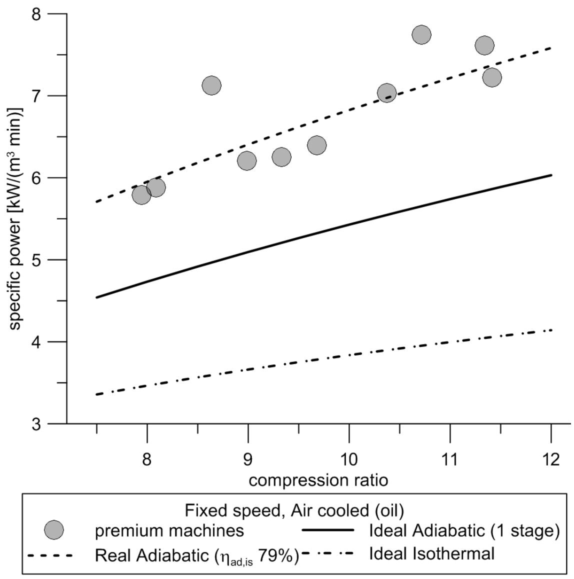


(b)

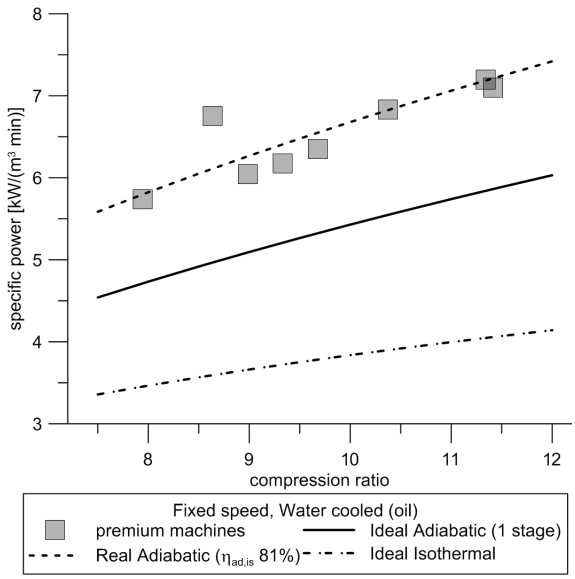
Fixed Speed, Water Cooled

Fig. 1: Market compressors performances (volumetric flow rate evaluated at 1 bar, 293.15 K)

machines and the ideal data is within a 1.1-1.4 $kW/(m^3/min)$ for the former, and between 1-1.3 $kW/(m^3/min)$, for the latter, with a compression ratio varying from 8 to 11. The difference between an ideal isothermal transformation and the adiabatic one is between 1.3 at 8 bar and 1.9 at 11 bar. As previously stated, in mean situations, the efficiency shift from the unit (i.e. the shift shown in Figure 2 between real data points and the adiabatic at η_{glob}=1) accounts almost entirely for mechanical, organic and electrical losses. However, the calculations focus on premium machines, in which these efficiencies have already reached an asymptotic (and optimum) value; as outlined, they still deserve scientific attention, but the potential improvement will start from a real situation which already is representative of premium machines. Consequently, the thermodynamic side is the most important to address to for further improvements and energy recovery. Curves in Figure 3 represent the specific power for an isothermal transformation (lower), an intercooled two stage adiabatic with a 30°C pinch point at the heat exchanger (middle) and a one stage adiabatic (upper), with the best achievable $\eta_{mech} \cdot \eta_{org} \cdot \eta_{el}$. It is evident how an ideal intercooled two stage recovers half the distance from the ideal isothermal which is the best reference transformation. The performance achievable with an



(a)



(b)

Fig. 2: Premium machines vs ideal adiabatic and isothermal

intercooled two stage compression is very promising. A quick comparison allows to quantify the gain in performances in percentages between 10-15% package power, with delivered pressure swinging from 6 and 15 bar. It is important to remark that these improvements underestimate the maximum achievable saving, if one considers that splitting in two stages the compression, the mechanical efficiency tends to increase. An interesting additional option is shown for SVRC: the dotted line represents the performances of a SVRC, when the oil is injected through a fine spray [20–23]. A mean 5% package power energy saving has additionally been found. As final remark, the attention must be reserved to the recovery of the heat discharged into environment by the oil. As observed, in any compressed air system involving positive displacement machines, only a 10-15% power consumption is compression related, with 85-90% package power lost and dissipated into heat. Most part of this (two thirds of the package power, from air cooling and friction) is removed by the oil during its stay within the machine and can be recovered after the compression process. In order not to lose its properties, the oil needs to be cooled down from $100°C$ at the compressor exit to 60-65$°C$, thus providing some low grade heat. These data suggest the importance of a proper energy recovery from this low grade thermal energy, and the great potential associated with its conversion into mechanical energy which could be directly substracted from the main compressor shaft. Recovery done with a unit based on an Organic Rankine Cycle (ORC) is very suitable and easily applicable in this industrial context. Recent studies demonstrate that this recovery can be done thanks to a new SVR Expander technology [11,12] and a global efficiency of 7-10% was measured when the oil was cooled down from 100 $°C$ to 65 $°C$. Recovering this energy and coupling it with the dual stage intercooled compression, a huge reduction in power consumption can be appreciated, with the curve referring to this situation closely approaching the isothermal one: a mean saving on the overall specific consumption is around 20-25% package power.

4. Conclusions

The Compressed Air Sector (CAS) is responsible for a relevant part of energy consumption, accounting for a mean 10% worldwide electricity needs. This ensures about the importance of the CAS issue when sustainability, in terms of energy saving and CO_2 emissions reduction, is in question. This is the main reason why the CAS is being re-thought as an area offering great opportunities for improvements. The compression transformation alone accounts for 10-15% package electricity consumption, thus attention has been paid to machine performances and to the definition of energy saving/recovery interventions.

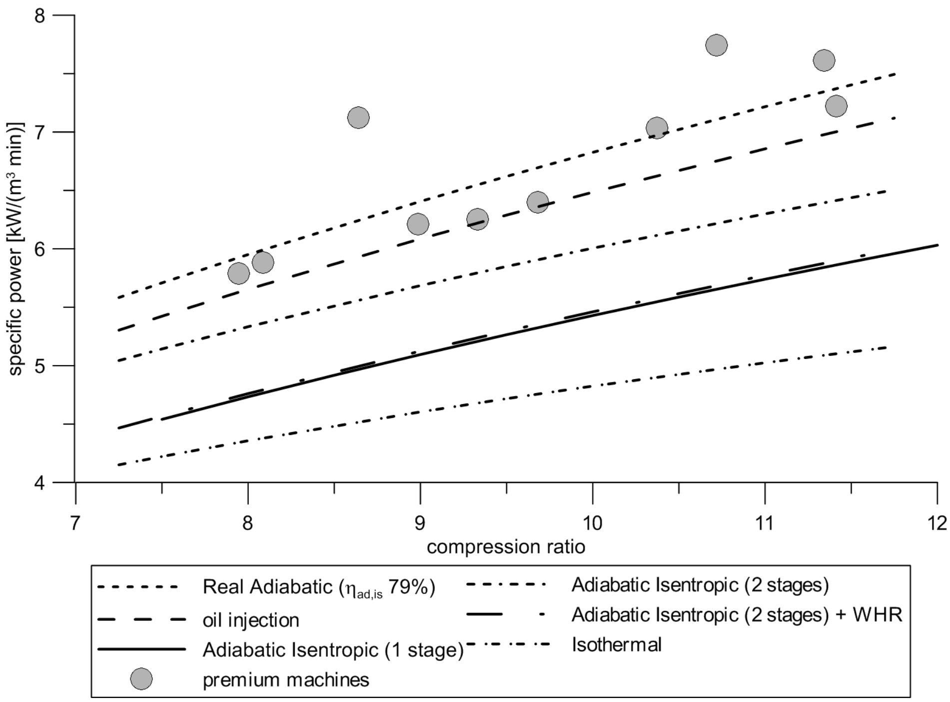


Fig. 3: SVRC - Fixed speed, Air cooled (approx. global efficiency=0.79)

The paper deals with compressor technology and it discusses the energy consumptions, on the basis of a comprehensive analysis of data for existing machines, mainly provided by the Compressed Air and Gas Institute (CAGI) for the US scenario. Data referring to different machine technologies, were processed to obtain consistency with fixed reference pressure levels and organized as a function of main operating parameters. A mean efficiency of 0.80 has been derived for premium machines on the market.

An analysis of the various terms of the overall compressor efficiency shows that such a number represents a consistent estimation of mechanical, organic and electrical terms contribution. Considering that both electrical and mechanical technology have reached a good level of development (even if improvements still remain possible) and that a reduced oil circulation accounts for a 5% package power, a great potential saving is possible on the thermodynamic side.

Present technology allows to consider a mean saving of 15% for an intercooled two stage compressor. The heat provided by the oil cooling is available at a temperature of about $100°C$, which allows a recovery into mechanical energy, by means of an Organic Rankine Cycle (ORC) based plant. A mean conversion efficiency, proved to be close to 7-10%, means very interesting margins for recovery, considering that the thermal power of the oil is a two thirds of the mechanical power absorbed by the compressor.

A safe estimation of the overall effect of a dual stage compression and waste heat recovery from the oil, fixes the saving/recovery potential at 20% package power. This benefit is achievable independently from the machine type, but opportunities for further improvement can still be catched according to the specific compressor considered: in SVRC the injection of a fine oil spray within the flow and downspeeding appear to be effective and easy to do. For Screw compressors, the greatest room for a compression power reduction comes from rotor improvement, while slightest or no margins are offered by downspeeding and oil circulation reduction. As a consequence, energy saving and energy recovery in the CAS have the potential to overcome the goals expressed by EC and World main Countries, in terms of energy and Carbon saving.

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References

[1] IEA, . World energy outlook. Tech. Rep.; 2013. Redrawing the Energy-Climate Map pp. 32-33.
[2] Tans, P., Keeling, R.. Noaa/esrl. 2013. URL: www.esrl.noaa.gov/gmd/ccgg/trends.
[3] EIA, . International energy outlook. Tech. Rep.; 2011. App. A-D, pp. 155-225.
[4] Radgen, P., Blaustein, E.. Compressed air systems in the european union. Stuttgart: LOG_X 2001;.
[5] PNEUROP, . Products in motor systems outside the scope of lot 30 and the regulation 640/209 on electric motors, in particular compressors, including small compressors, and their possible drives. Tech. Rep.; 2013. ENER Lot 31.
[6] CAGI, . Rotary compressor performance verification program. 2013. URL: www.cagi.org/performance-verification/data-sheets.
[7] Cipollone, R.. Sliding vane rotary compressor technology and energy saving. Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering 2014;doi:10.1177/0954408914546146.
[8] Cipollone, R., Vittorini, D.. Energy saving potential in existing compressors. In: 22nd International Compressor Engineering Conference. 2014,.
[9] Cipollone, R., Bianchi, G., Contaldi, G.. Ottimizzazione energetica di compressori a palette. In: Proceedings of the 67th Conference of the Italian Thermal Machines Engineering Association. ISBN 978-88-907676-0-9; 2012, p. 144–153. doi:10.13140/2.1.4835.7442.
[10] Bianchi, G., Cipollone, R.. Theoretical modeling and experimental investigations for the improvement of the mechanical efficiency in sliding vane rotary compressors. Applied Energy 2014;doi:10.1016/j.apenergy.2014.12.055.
[11] Cipollone, R., Bianchi, G., Battista, D.D., Contaldi, G., Murgia, S.. Mechanical energy recovery from low grade thermal energy sources. Energy Procedia 2014;45(0):121 – 130. doi:http://dx.doi.org/10.1016/j.egypro.2014.01.014.
[12] Cipollone, R., Contaldi, G., Bianchi, G., Murgia, S.. Energy recovery using sliding vane rotary expanders. In: Engineers, I.o.M., editor. 8th International Conference on Compressors and their Systems. Woodhead Publishing. ISBN 978-1-78242-169-6; 2013, p. 183 – 194. doi:http://dx.doi.org/10.1533/9781782421702.3.183.
[13] Cipollone, R., Contaldi, G., Sciarretta, A., Tufano, R.. A comprehensive model of a sliding vane rotary compressor system. In: Institution of Mechanical Engineers - 4th International Conference on Compressors and Their Systems. 2005, p. 195–205.
[14] Stosic, N., Smith, I., Kovacevic, A., Mujic, E.. Geometry of screw compressor rotors and their tools. Journal of Zhejiang University SCIENCE A 2011;12(4):310–326. doi:10.1631/jzus.A1000393.
[15] Stosic, N., Smith, I., Kovacevic, A., Kim, J.U., Park, J.. Improving screw compressor performance. In: Institution of Mechanical Engineers - 4th International Conference on Compressors and Their Systems. 2005, p. 13–22.
[16] Bianchi, G., Cipollone, R.. Friction power modeling and measurements in sliding vane rotary compressors. Applied Thermal Engineering in press;.
[17] Stosic, N., Smith, I., Kovacevic, A.. Optimisation of screw compressors. Applied Thermal Engineering 2003;23(10):1177 – 1195. doi:http://dx.doi.org/10.1016/S1359-4311(03)00059-0.
[18] Stosic, N., Smith, I., Kovacevic, A.. Rotor interference as a criterion for screw compressor design. Journal of Engineering Design 2003;14(2):209–220. doi:10.1080/0954482031000091518.
[19] Cipollone, R., Valenti, G., Bianchi, G., Murgia, S., Contaldi, G., Calvi, T.. Energy saving in sliding vane rotary compressors. In: Engineers, I.o.M., editor. 8th International Conference on Compressors and their Systems. Woodhead Publishing. ISBN 978-1-78242-169-6; 2013, p. 173 – 181. doi:http://dx.doi.org/10.1533/9781782421702.3.173.
[20] Cipollone, R., Bianchi, G., Contaldi, G.. Sliding vane rotary compressor energy optimization. In: ASME International Mechanical Engineering Congress and Exposition (IMECE); vol. 6. 2012, p. 69–80. doi:10.1115/IMECE2012-85955.
[21] Bianchi, G., Cipollone, R., Murgia, S., Contaldi, G.. Performance enhancement in sliding vane rotary compressors through a sprayed oil injection technology. In: 22nd International Compressor Engineering Conference. 2014,.
[22] Cipollone, R., Valenti, G., Bianchi, G., Murgia, S., Contaldi, G., Calvi, T.. Energy saving in sliding vane rotary compressors using pressure swirl oil atomizers. Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering 2014;doi:10.1177/0954408914550356.
[23] Bianchi, G., Cipollone, R., Murgia, S., Contaldi, G.. Development of an internal air cooling sprayed oil injection technique for the energy saving in sliding vane rotary compressors through theoretical and experimental methodologies. International Journal of Refrigeration 2014;doi:10.1016/j.ijrefrig.2014.12.020.